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MECHANICAL ENGINEERING NOTE 367

ESTIMATION OF CABIN HEAT LOADING IN THE SEA KING HELICOPTER AND EFFECTIVENESS OF POSSIBLE MODIFICATIONS TO IMPROVE CREW THERMAL ENVIRONMENT

by

BRIAN REBBECHI

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SUMMARY

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16. *ABSTRACT*

Measurements of temperatures and airflows have been made in a Sea King helicopter to determine the parameters influencing the thermal environment. From these measurements a heat flow model is developed. The effectiveness of minor modifications to partially alleviate the hot conditions is discussed, and four proposals are made. However, it is confirmed that the only way to bring the cabin thermal environment to a completely acceptable level is by the use of refrigerated air supplies. One specific refrigeration proposal, that of a Normalair-Garrett Commando cooling unit, is assessed and found to provide a marginally acceptable crew environment.

CONTENTS

NOMENCLATURE

A_e	Area of fuselage exposed to hot exhaust gas (m^2)
A_f	Area of floor (m^2)
C	Arbitrary constant
C'	Arbitrary constant
C_p	Specific heat of air (kJ/kg °C)
ET	Effective Temperature (°C)
H_a	Known heat loadings (kW)
H_{cabin}	Heat transfer into cabin (kW)
H_d	Heat input due to hot air (kW)
H_e	Cockpit and cabin electrical load (kW)
H_{eng}	Heat input to cabin due to engine (kW)
H_m	Metabolic heat load from occupants (kW)
h_c	Heat removed by cooling unit air (kW)
h_f	Heat flow through floor (kW)
h_{wall}	Heat flow through cabin walls. (kW)
K_e	Overall thermal conductivity of fuselage (kW/°C)
k_f	Thermal conductivity of floor (kW/°C)
m_c	Cooling unit air mass flow (kg/s)
m_d	Hot air mass flow (kg/s)
m_f	Cabin ventilation air mass flow (kg/s)
m_l	Leakage inflow of ambient air (kg/s)
m_r	Ram-air mass flow (kg/s)
q_f	Heat flow through floor (kW)
q_t	Total heat flow through fuselage (kW)
RH	Relative humidity (%)
T_a	Ambient air temperature (°C)
T_c	Cabin air temperature (°C)
T_d	Hot air temperature (°C)
T_{db}	Dry bulb air temperature (°C)
T_f	Floor temperature (°C)
T_{fuel}	Fuel temperature (°C)
T_g	Black globe temperature (°C)
T_{in}	Cabin ventilation air temperature at entry to cabin (°C)
T_{inc}	Cooling unit air temperature at cabin entry (°C)
T_{wb}	Psychrometric wet bulb temperature (°C)
T'_{wb}	Wet bulb temperature of naturally convected thermometer (°C)
δT_c	Temperature difference between cabin air and ambient air temperature (°C)
δT_e	Temperature increment of exhaust gas heated air above ambient air (°C)
δT_r	Stagnation temperature rise of ram-air (°C)
δT_{vent}	Temperature rise of air due to cabin ventilation fan (°C)
t	Time (s)
V	Volume of cockpit and cabin (m^3)
V_a	Air velocity (m/s)
WBGT	Wet bulb globe temperature (°C)
W_{in}	Humidity ratio of cabin inlet air (kg moisture/kg dry air)
W_a	Humidity ratio of ambient air (kg moisture/kg dry air)
w_c	Humidity ratio of cabin air (kg moisture/kg dry air)

1. INTRODUCTION

Sea King Mark 50 helicopters currently in service with the Royal Australian Navy are experiencing problems of noise, vibration and excessive cabin temperatures during normal ASW operations. These Laboratories have been asked to determine the magnitude of these problems and to assess the effectiveness of various methods of alleviation.

This note is concerned with the cabin thermal environment which has been found to have adverse effects on crew performance, and with possible modifications to improve the situation. Calculations for this note are based on data gathered by the author during flights at Nowra in February and June 1977. Further data is being obtained during Service operation over a period of twelve months of a Sea King instrumented to measure the cabin environment in an unmodified aircraft. This aircraft is operating from HMAS *Melbourne* in a wide range of climatic conditions. While the aircraft is instrumented specifically to measure factors affecting crew comfort and efficiency, it is expected that some of the readings will serve as checks on the present computations.

The present report is in two main sections, the first of which analyses the current problem and discusses ways in which the hot conditions could be alleviated by minor modifications. The second section analyses in detail the heat balance in the cabin, and assesses the performance of a proposed air-cycle cooling system.

No great accuracy is claimed for the analysis and assessment because of the minimal instrumentation used in the tests. It is, however, considered that the conclusions reached would not be affected in a practical way by elaborating the measurements.

2. ENVIRONMENT PARAMETERS

2.1 The External Environment

The values used here to assess the maximum cabin temperatures and humidities in this note are the RAAF atmospheric environment (for temperature) and the AvP 970 atmospheric environment (Ministry of Aviation, 1960—Ref. 1) for humidities.

The maximum sea level temperature is therefore taken as 43°C, and the design maximum humidity limits are as shown in Figure 2.1.

2.2 The Cabin Environment

Numerous indices are used to bring together the environmental parameters which influence heat dissipation from a human subject, the principal parameters being the dry bulb temperature (T_{db}), wet bulb temperature (T_{wb}), radiant temperature of surroundings, and air velocity. Two of these indices are referred to here.

2.2.1 Wet Bulb Globe Temperature

The wet bulb globe temperature (WBGT) takes account of incident thermal radiation, ambient temperature and humidity. It is defined (Kerslake, 1972—Ref. 2) as

$$\mathrm{WBGT} = 0{\cdot}7\, T_{wb} + 0{\cdot}3\, T_g \qquad (2.1)$$

where T_{wb} = psychrometric wet bulb temperature (°C),

T_g = temperature of a 150 mm black globe (°C).

The Aircraft Research and Development Unit (1969*a*, 1969*b*—Refs 3, 4) and the Institute of Aviation Medicine, RAAF (1976—Ref. 5) have concluded that an upper limit of 28° WBGT is necessary if physiological stress and impairment of performance of pilot and aircrew are to be avoided.

An alternative expression is

$$\mathrm{WBGT} = 0{\cdot}7\, T'_{wb} + 0{\cdot}1\, T_a + 0{\cdot}2\, T_g, \qquad (2.2)$$

where T'_{wb} is the naturally convected wet bulb temperature, and T_a is the dry bulb temperature. This alternative is often more suited to field measurement of WBGT than equation (2.1).

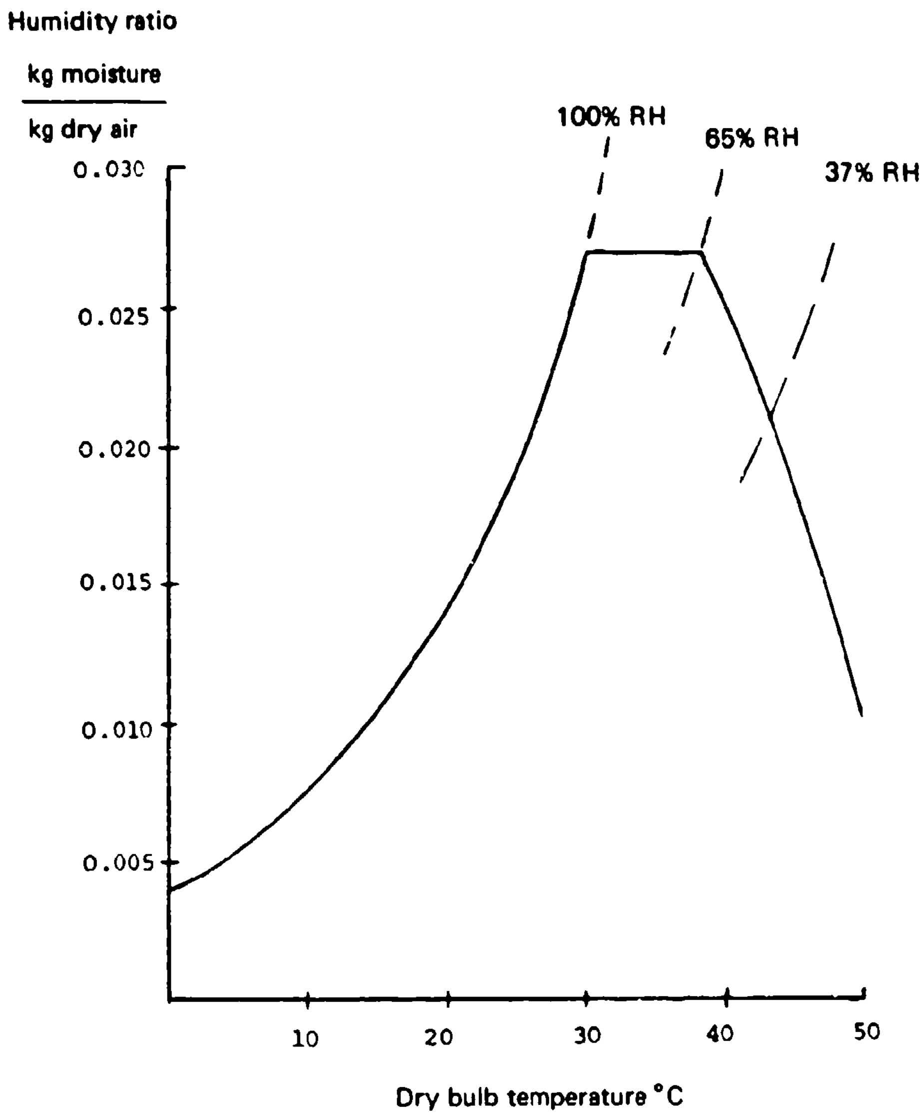

FIG. 2.1 DESIGN MAXIMUM HUMIDITY LIMIT RELATED TO TEMPERATURE FOR SEA-LEVEL (Reproduced in modified form from Ministry of Aviation (1970) – Ref. 1)

This limit is not a 'soft' criterion; it is specified for acclimatised, fit, well-hydrated aircrew with no exposure to heat stress on that day. WBGT values of 35°C are sufficient to induce a situation of near collapse in a subject after 1 hour; this was demonstrated during Nomad aircraft trials conducted by the Institute of Aviation Medicine (1976—Ref. 5).

Variations in clothing will affect tolerance to a particular WBGT; in this regard the RAN 'Mae West' appears to be of somewhat heavier construction than the RAAF version, and so may lower the acceptable WBGT.

It should not be assumed that an environment of 28°C WBGT is a desirable maximum; continuous sweating of a subject will occur at this temperature, resulting in the soaking of some areas of clothing with perspiration, and the possibility of free perspiration obscuring vision.

2.2.2 Effective Temperature

The effective temperature (ET) is the temperature of still, saturated air which would give rise to an equivalent sensation of thermal comfort to that experienced with air movement. It is derived from a nomogram (see Fig. 2.2) and is for subjects wearing normal indoor clothing. This index is included here because of its widespread use; studies of mental performance decrement tend to be carried out in the ET scale. For example, Figure 2.3 has been reproduced (in modified form) from Iampietro (1970—Ref. 6).

2.2.3 Comparison of WBGT and ET

The WBGT is an easier index to measure than the ET, as no measurement of air movement is required. Because of its simplicity, it is widely used in military applications. The two indices are not directly comparable numerically; however, for a low air velocity (around 0·5 m/s) such as is found in the Sea King cabin, and an ET of 28–30°C, the weighting of the wet bulb and globe temperatures for the two indices is similar. For example, at a WBGT of 28°C, where $T_g = 41$°C, $T_a = 35$°C, $T_{wb} = 23 \cdot 3$°C (relative humidity 38%), and an air velocity of 0·5 m/s, the ET is 29·4°C.

3. THE CURRENT SEA KING HEATING PROBLEM

In the course of measurements made primarily to develop a heat flow model for the Sea King cabin, two factors became evident which appear unnecessarily to exacerbate the current heating problems. These factors, namely a hot air flow into the cabin and a substantial temperature rise across the cabin ventilation fan, are discussed in Sections 3.1 and 3.3. The cabin airflow patterns are discussed in Section 3.2.

3.1 Entry of Hot Air to the Cabin

Entry of hot air to the cabin was found to occur through an area above and behind the pilot's and co-pilot's head (between the cabin lining and fuselage skin), and from an area behind the pilot's seat which houses the cyclic pitch control hydraulic servomechanisms (this latter area is commonly termed the 'broom cupboard'). This hot air flow was found to originate from a hot region around the main transmission gearbox; entry into the cabin was along a duct housing the cyclic pitch control rods (Fig. 3.1). Local heating by hydraulics within the cabin does not contribute significantly to the heating of this airflow.

This hot air enters only when either or both of the front vent windows are open (Figs 3.2, 3.3, 3.4) in forward flight. During the hover it was noted that the airflow in this region decreased by a factor of three, although the direction of flow was not observed. During forward flight with the front vent windows closed there was a small flow of cabin air (0·05 kg/s) outwards along the cyclic pitch control duct. In this case the 'broom cupboard' temperature was approximately equal to the cabin temperature—thus confirming that heating of the 'broom cupboard' is due to the flow of hot air through it, and not to local heating by hydraulics.

For the forward flight case the airflows and heating effect due to this entry of hot air are shown in Table 3.1. The cabin air ventilation fan was switched on during these tests.

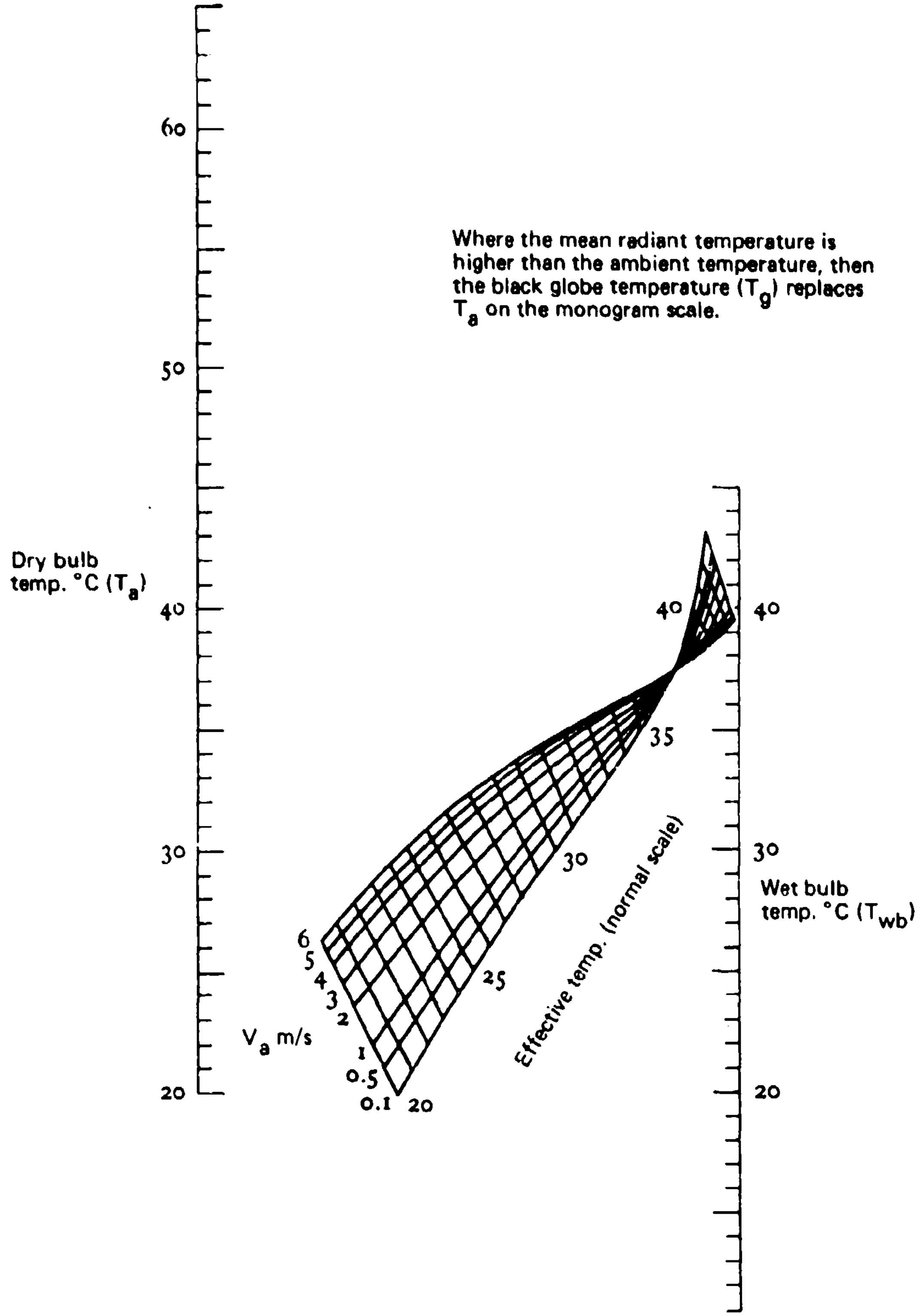

FIG. 2.2 NOMOGRAM FOR DETERMINATION OF EFFECTIVE TEMPERATURE (Reproduced from Kerslake (1972) – Ref. 2)

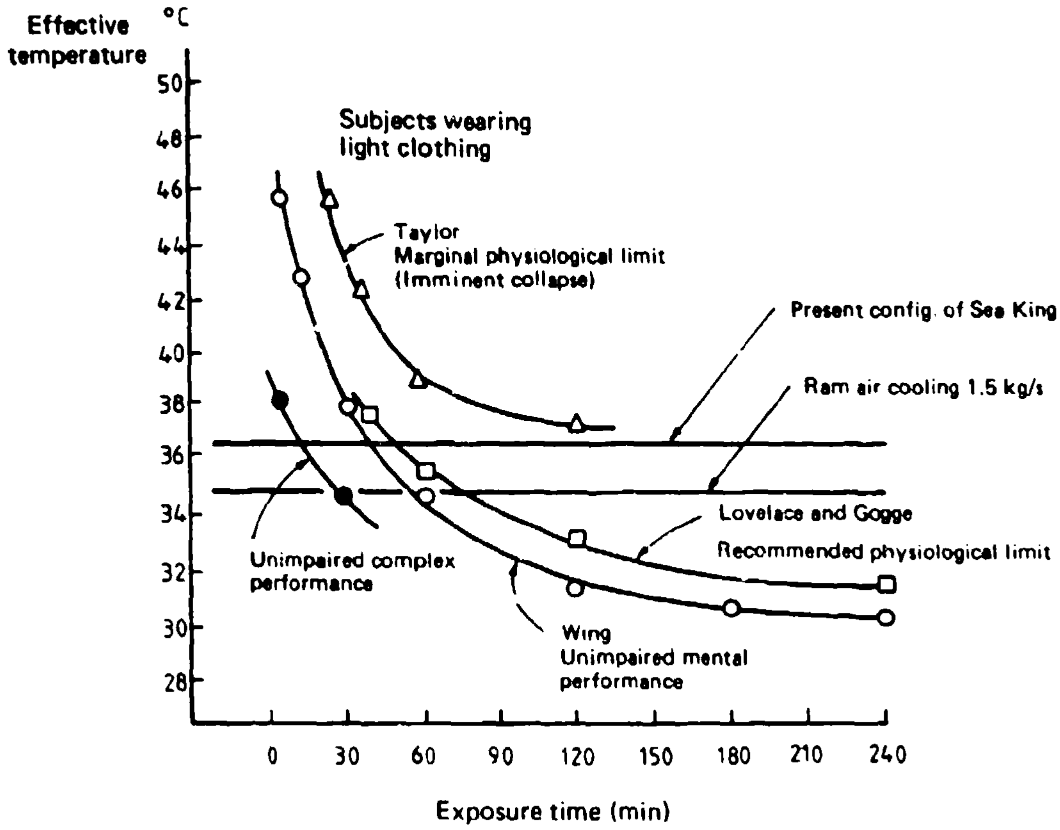

FIG. 2.3 PHYSIOLOGICAL AND PERFORMANCE LIMITS IN HOT ENVIRONMENTS
(Modified from Iampietro (1970) – Ref. 6)

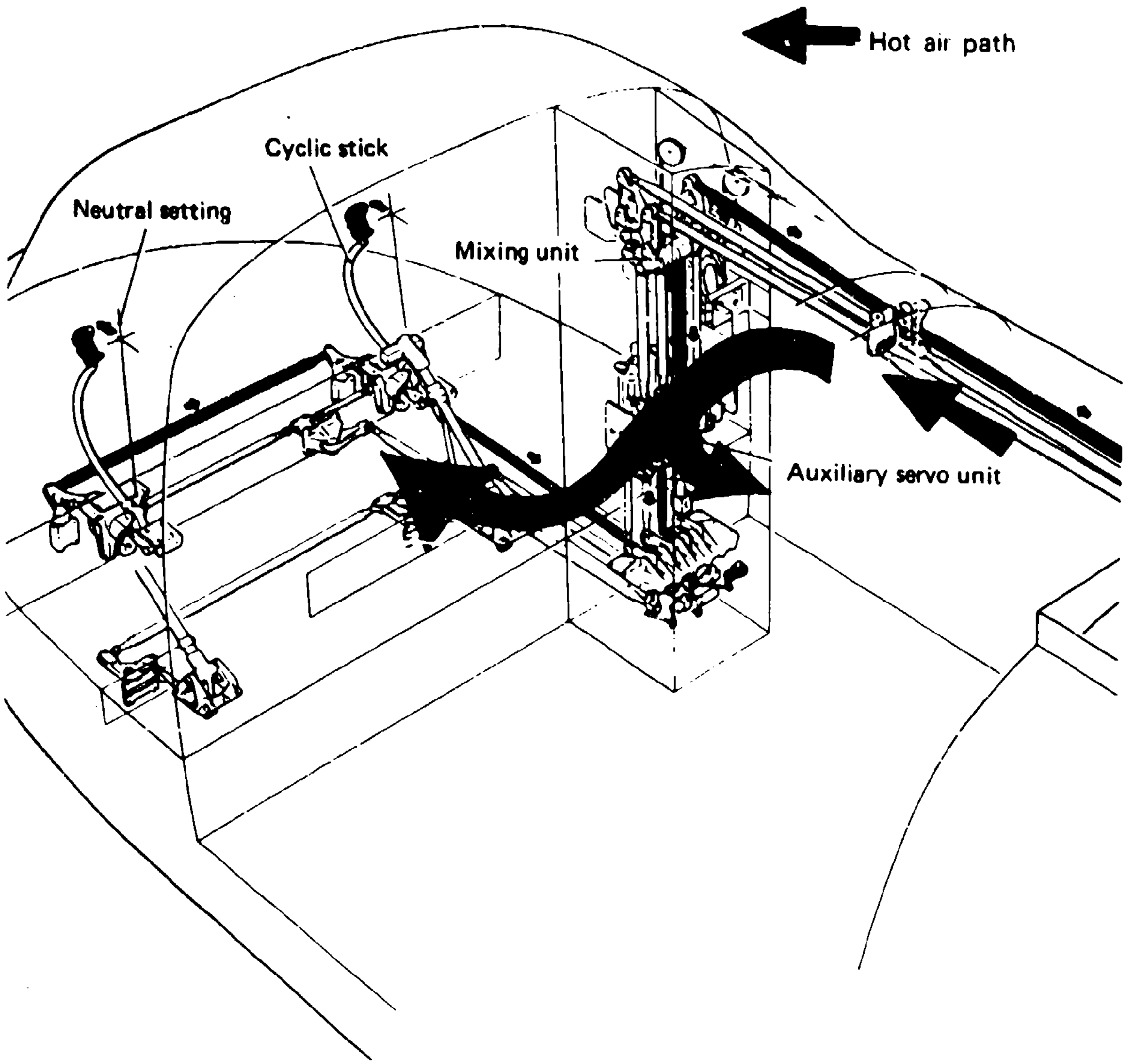

FIG. 3.1 ENTRY PATH OF HOT AIR TO CABIN.

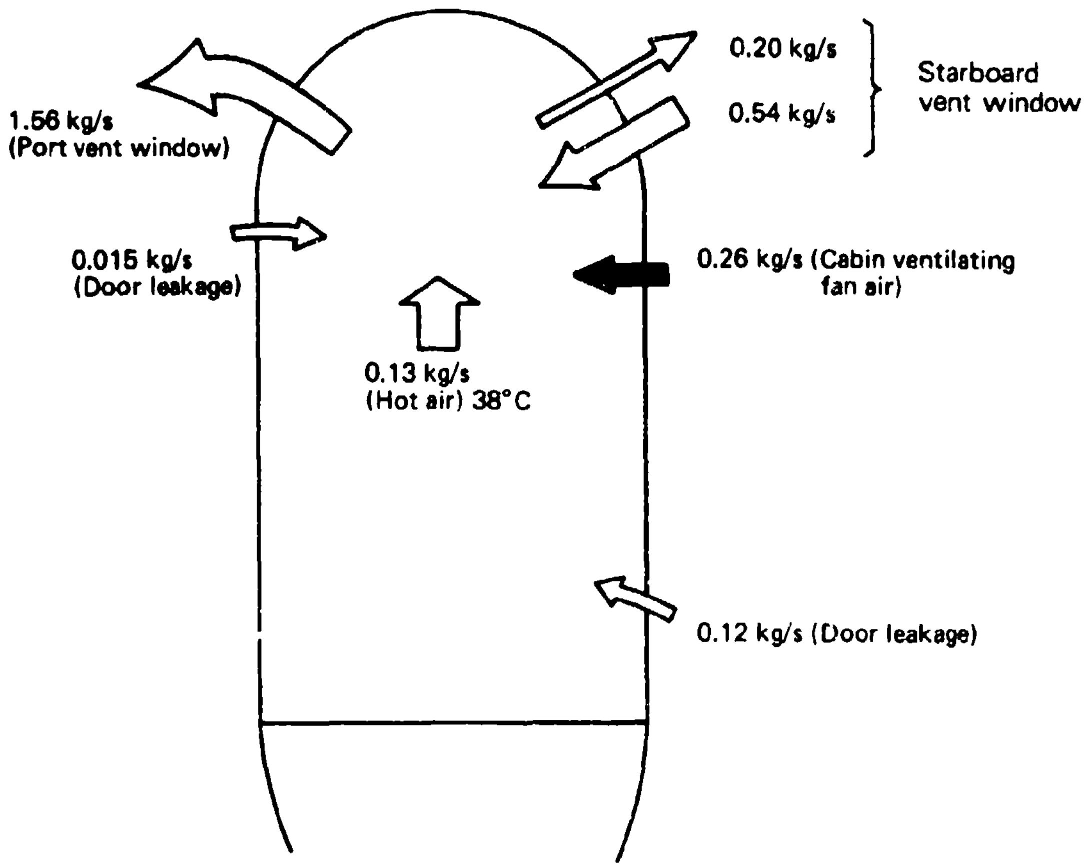

FIG. 3.2 CABIN AIRFLOW PATTERNS IN FORWARD FLIGHT WITH BOTH FRONT VENT WINDOWS OPEN.

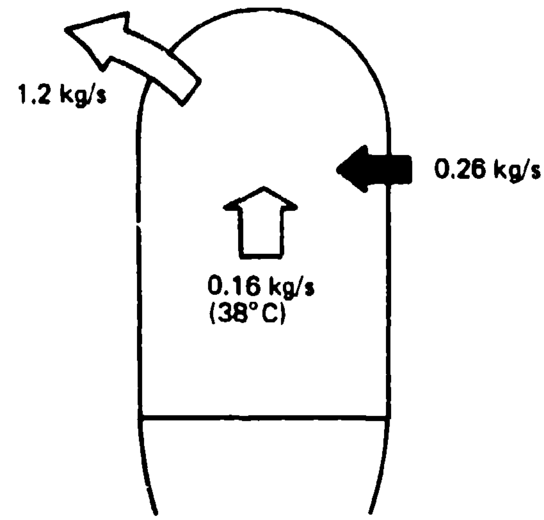

FIG. 3.3 CABIN AIRFLOW PATTERNS IN FORWARD FLIGHT WITH PORT WINDOW OPEN.

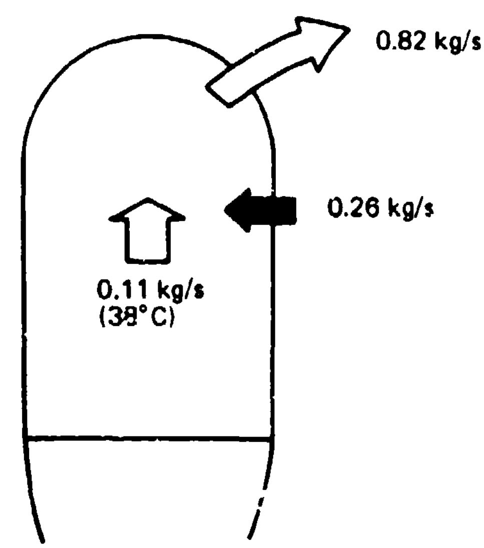

FIG. 3.4 CABIN AIRFLOW PATTERNS IN FORWARD FLIGHT WITH STARBOARD WINDOW OPEN.

TABLE 3.1

Heating effect and mass flows of air entering cabin through the 'broom-cupboard' and area above pilot's head

Window position	Airspeed (knots)	Mass flow of hot air (kg/s)	Temp. of hot air (°C)	Ambient outside temp. (°C)	Heating effect on cabin (kW)	Flow direction
Both front vents closed	80			15	Nil	Out of cabin
Both front vent windows open	100	0·13	38	16	1·56	Into cabin
Starboard window only open	100	0·11	38	16	approx. 1·3	Into cabin
Port window only open	100	0·16	38	16	approx. 1·9	Into cabin

It is thus apparent from Table 3.1 that a considerable heat flow (up to 1·9 kW) into the cabin, exists due to the entry of this hot air. Moreover, it exists in the worst situation, namely where the pilots have opened the front windows to obtain relief from the hot cabin conditions.

3.2 Cabin Airflow Patterns

The cabin airflow patterns are illustrated in Figures 3.2, 3.3 and 3.4. The airflows are approximate only, for the following reasons;

(*a*) airflows will be influenced by a skewness of the aircraft to the line of flight;

(*b*) measurements were made with a hand-held anemometer;

(*c*) the 'fit' of the doors varied on successive flights and between different aircraft.

It is apparent from Figures 3.2, 3.3 and 3.4 that the *measurements* of the total inflows of air do not equal the total outflows. This is because, firstly, only the major detected areas of flow were measured, and secondly because of possible errors in measurement. However, these errors are not considered to invalidate the conclusions drawn in this note.

3.3 Temperature Rise Across Cabin Ventilating Fan

The temperature rise of the air passing through the cabin ventilating fan was found to range from 6–8°C, the majority of readings being 7·5°C. The inlet and outlet temperatures were measured at the locations as shown in Figure 3.5. The temperature rise of 7·5°C, associated with a mass flow through the fan (measured by a hand-held anemometer) of 0·26 kg/s (35 lbm/min) implies a power input to the air of 1·95 kW. The fan has a nominal rating of 1·4 kVA, according to load sheets for the Sea King. Thus the fan motor may be overloaded—it is unlikely that the ventilation air mass flow measurement is greatly in error. Another source (Directorate of Naval Aircraft Engineering File Ref. N1520/7/73, 21/2/77, Appendix E—Ref. 7) quotes a mass flow of 0·29 kg/s (40 lbm/min), which would imply an even larger power input to the cooling air. The inlet air to the fan was 2°C above the ambient outside air temperature, presumably due to the forward airspeed of 50 m/s (100 knots).

4. PARTIAL ALLEVIATION OF SEA KING HEATING PROBLEM

There are several short-term ways in which significant reductions in Sea King cabin temperatures could be made. These reductions will not bring the cabin environment to a fully acceptable level as defined in Section 2. However, the improvements could conceivably be incorporated before the 1977–78 Australian summer, and in some cases would entail little, if any, structural modification. The various possibilities are outlined in the following Sections 4.1 to 4.4, and are summarised in Section 4.5.

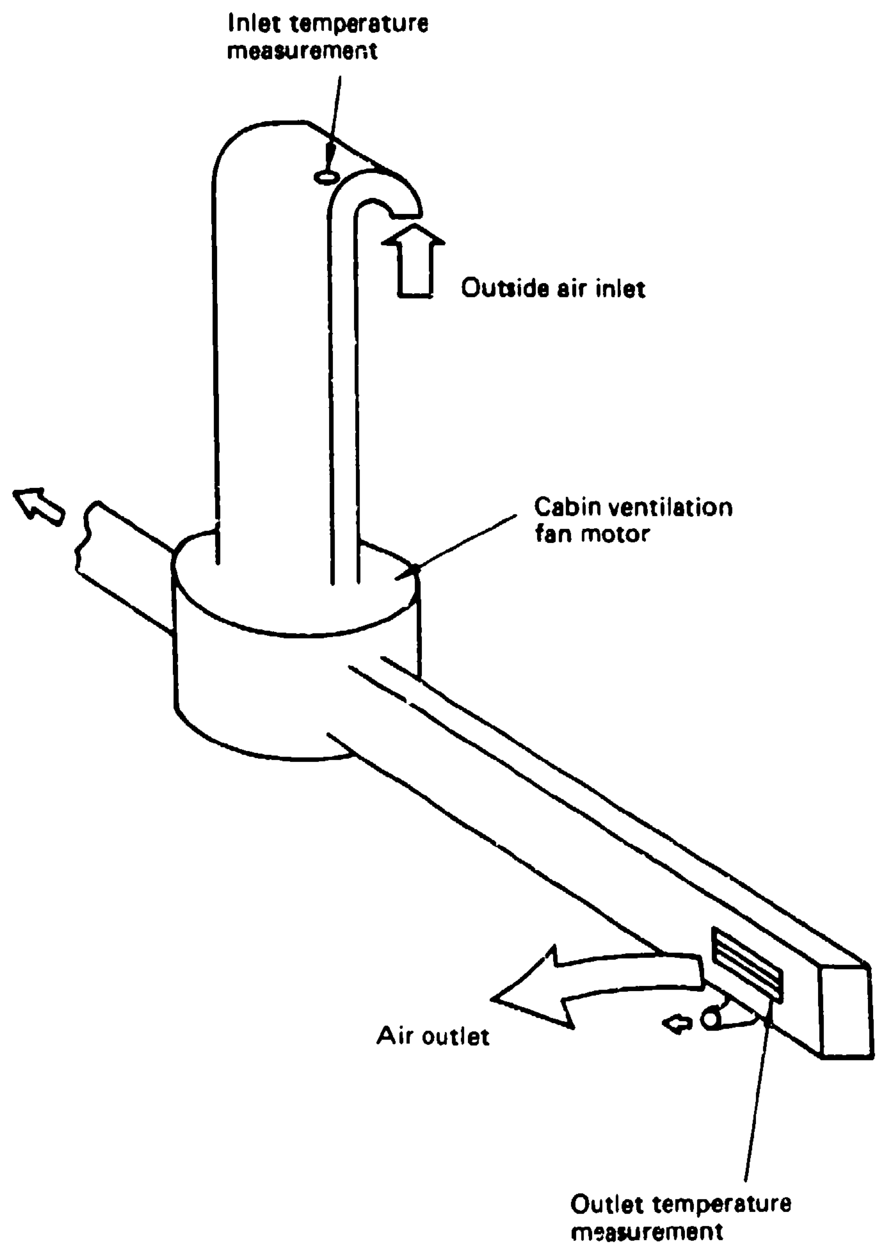

FIG. 3.5 MEASUREMENT POINTS FOR TEMPERATURE RISE OF AIR PASSING THROUGH THE CABIN VENTILATION AIR FAN.

4.1 Prevention of Hot-Air Entry

The entry of hot air to the cabin could be prevented by a relatively small modification, whereby the flow of air along the cyclic control duct is stopped. Means could be employed such as rubber gaiters or sliding plates (as in Figs 4.1 and 4.2), providing allowance is made for lateral displacement of the control rods.

The improvement resulting from this modification would be considerable—of the order of 1·5–2·0 kW reduction in heat input to the cabin under the ambient conditions experienced at Nowra (see Table 3.1).

It can be shown from the heat balance equation (A1.19, Appendix 1) that this heat input lies between 31% and 22% of all other heat flows into the cabin, depending on whether the sonar is in operation.

4.2 Minimising the Effect of Air Temperature Rise Across Cabin Ventilating Air Fan

It is clear from Figures 3.2 to 3.4 that depending on window openings and door leakages, airflows into and out of the cabin may be considerably greater than that through the cabin air fan. However, as there is a considerable temperature rise in the fan, this so-called ventilating air in fact contributes 2·0 kW heat input to the cabin referred to ambient air temperature as a datum. It can be seen from Figure 5.1 that the same airflow at ambient temperature would effect a reduction of 2·5°C in cabin temperature, when the cabin vent windows are open.

Means of achieving this have not been examined in detail. However, it might be feasible to operate the present fan as an exhauster, passing hot cabin air out through the present inlet. If this were done care would be needed to provide alternative low pressure loss air inlets to the cabin so arranged that pick-up of hot air or exhaust gases is avoided at all conditions.

4.3 Ram Air Cooling of Cockpit and Cabin

Cooling by ram air would overcome the disadvantages of the fan temperature rise discussed in the previous Section 4.2, and the air is potentially available in much greater quantities, at a temperature only slightly (1·5°C stagnation temperature rise) above ambient.

Because of the variable flight configurations (forward flight and hover), Sikorsky have manufactured a rotating scoop which is fitted in place of the starboard cabin window of SH-3 aircraft. There are no documented results available regarding the performance of this scoop, although adverse verbal reports are noted (Directorate of Naval Aircraft Engineering, File Ref. N1520/7/73, 21/2/77, Appendix C—Ref. 7).

Provided that adequate air exit points, at a lower pressure than the scoop inlet can be incorporated, there should be a through-flow of air in both the pilot and observer/sonar operator areas, and this scoop will bring about a considerable reduction in the cockpit–cabin air temperature.

For example, from Figure 5.1, for the situation where the heat loading H_a is 5·5 kW (full solar radiation, sonar on), a ram air flow of 1·5 kg/s (198 lbm/min) will reduce the cabin temperature as shown in Table 4.1.

TABLE 4.1
Effect of ram-air cooling on cabin temperature rise above ambient temperature

Temperature rise above ambient (°C) with sonar in operation		
Present configuration cabin ventilating air on		Ram-air cooling 1·5 kg/s
Vent windows open	Vent windows closed	Cabin ventilating fan on
12°C	16·5°C	5·6°C

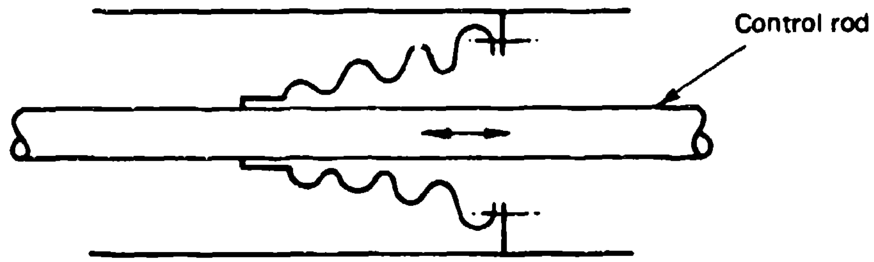

FIG. 4.1 PREVENTION OF HOT AIR ENTRY BY RUBBER GAITER.

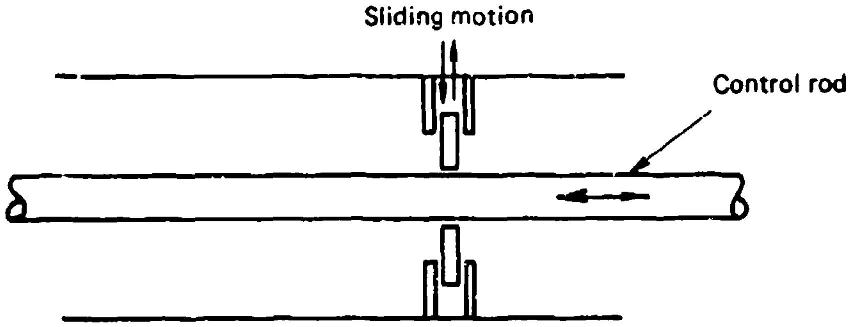

FIG. 4.2 PREVENTION OF HOT AIR ENTRY BY SLIDING PLATE.

4.4 Reduction in Heat Load Imposed by Avionics on the Cabin

The electrical load in the cabin has been taken, from load sheets, as 0·5 kW exclusive of the sonar equipment; the electrical load of the sonar equipment inside the cabin being 2·5 kW. At present, this heat load passes directly into the cabin atmosphere and thus contributes substantially to the cabin bulk air temperature.

A significant improvement to the cockpit/cabin environment would be achieved if air used to cool the sonar equipment could be ducted directly overboard; in this case the greatly increased electrical loads envisaged for the updated Sea King could be accommodated without increasing the cabin cooling problem. A separate cooling fan supplying air at ambient temperature would be best for this purpose.

New ASW electronic equipment is proposed for an updated Sea King, which would considerably increase the electrical load. It appears that the temperature limitations of this new equipment have been misinterpreted; the temperature limitation of 45°C referred to in Appendices to Directorate of Naval Aircraft Engineering File Ref. N1520/7/73, 21/2/77—Ref. 7, and earlier correspondence on this file, refers to the air *entering* the equipment, not that *leaving*. Calculations of cooling air *exit* temperatures, based on recommended cooling airflows, for sections of the equipment for which design figures are available, show that a maximum of 60°C is allowable. This greatly simplifies the cooling problem for this equipment.

4.5 Summary

Any one of the suggestions of Sections 4.1 to 4.4 will offer a significant decrease in cabin temperatures of the Sea King. These proposals could be adopted simultaneously.

Of the foregoing proposals, only ram air scoops are known to have been tested, though little information, and that unfavourable, is available regarding these tests. On the evidence available, we find it difficult to believe that these scoops would be ineffective in high outside ambient temperatures, as suggested in Directorate of Naval Aircraft Engineering File Ref. N1520/7/73, 22/2/77, Appendix C—Ref. 7.

Their application would, however, need to be carefully planned, as outlined in Section 4.3.

These proposals for a partial alleviation of the Sea King heating problem should not be construed as *satisfactory* solutions to the problem. The best possible improvement to the Sea King environment, without using refrigerated air supplies, would reduce the cabin to a temperature of 4·5°C above ambient.

Achieving this would involve suppression of hot air inflows, ducting sonar ventilation air overboard, and supplying at least 1·5 kg/s ram cooling air. On the basis of the most adverse atmospheric conditions (see Section 2.1) the resulting WBGT would be 37·7°C. This compares with a value of 40·8°C computed for the unmodified aircraft operated with both vent windows open.

These WBGTs are approximately equivalent to Effective Temperatures of 34·8°C and 36·5°C respectively, assuming a local air velocity in the cabin of 0·5 m/s. Thus, from Figure 2.3, and the work of the Institute of Aviation Medicine, temperatures by all criteria remain too high for efficient crew performance. This is largely because ram air cooling, while reducing the dry bulb temperature in the cabin, does not reduce the moisture content of the air, and hence only slightly reduces the wet bulb temperature. Thus, heat dissipation by sweating, which is the main physiological cooling mechanism for humans at high temperatures, is virtually unchanged.

5. HEAT BALANCE OF SEA KING

The heat balance of the Sea King was analysed primarily to assess the effectiveness of possible cooling units, though it has also been of use in assessing the usefulness of minor modifications to the aircraft. The computations are contained in Appendix 1 to this note; details of the final balance equation are given below.

The heat balance is for steady state conditions only. The transient case was not analysed because of the limited time available; it is considered, however, that large transients, where the aircraft has to be cooled from a "hot-soak", should be avoided where possible by shading the aircraft.

5.1 Known Internal Heat Loadings

These are:

Human metabolic heat (4 persons)	0·5 kW
Cockpit and cabin avionics	0·5 kW
Sonar (if on)	2·5 kW

5.2 Solar Radiation

The most significant source of external heating is solar radiation passing through the transparencies. Assuming a maximum solar intensity of 1000 W/m^2, average transparency transmittance of 0·83, and plan projected area of 2·4 m^2, then the heat load due to solar radiation is 2·0 kW.

5.3 Heat Balance Equations

From Appendix 1 the heat balance equation for the Sea King in its current configuration, is:

$$H_a = \delta T_c\,(0{\cdot}17 + m_l + m_f + m_r) - \delta T_{\text{vent}}\, m_f - \delta T_r\, m_r - 0{\cdot}98, \qquad \text{(A1.20)}$$

where H_a = heat loading (metabolic + avionic + solar) (kW);

δT_c = temperature difference between outside air and cabin ($T_c - T_a$) (°C);

m_l = leakage inflow of ambient air (kg/s);

m_f = ventilation fan mass flow (kg/s);

m_r = ram air mass flow (kg/s);

δT_{vent} = temperature rise of air through ventilation fan (°C);

δT_r = stagnation temperature rise of ram air (°C).

For the current configuration, with front vent windows closed and cabin air on, equation (A1.20) can be simplified to

$$H_a = 0{\cdot}54\,\delta T_c - 3{\cdot}45. \qquad \text{(A1.18)}$$

Equation (A1.20) can be presented graphically as in Figure 5.1, where the effects on the cabin temperature of various changes in cabin airflows are shown.

An estimate is made in Figure 5.1 of the heat balance for the current aircraft with both front vent windows open. This estimate is based on a reported in-flight maximum temperature rise above ambient of 12°C (minutes of a meeting on the Sea King environmental survey, 20/7/76—Ref. 8), presumably with sonar equipment switched on. Sufficient data is not yet available from the instrumented Sea King (Number 07) for this to be confirmed.

6. REVIEW OF CABIN AIR-CONDITIONING ALTERNATIVES

From the heat balance equation (A1.20) the cabin environment resulting from a particular cooling unit can be assessed. One proposal is for a Normalair-Garrett Commando cooling unit to be installed in the Sea King. The effectiveness of this unit is assessed below.

6.1 NGL Commando Cooling System

The performance of this system (Directorate of Naval Aircraft Engineering File Ref. N1520/7/73, 21/2/77, Appendix E—Ref. 7) is given as an outlet temperature of 4°C at a mass flow of 0·11 kg/s (15 lbm/min) for ambient conditions of 35°C and moisture content of 0·026 kg moisture/kg dry air. With dry air, this performance should quite easily be achieved at the RAAF atmospheric temperature of 43°C, which is taken as an upper limit for these calculations. With moist air, there is however, a substantial loss of cooling due to the heat of condensation of the water vapour. Without further data, an accurate estimate of the performance of the cooling unit with moist air cannot be made. For this reason, performance estimates are given below for an ambient temperature of 43°C, assuming alternatively

(*a*) that air leaves the cooling unit at 4°C, and

(*b*) that there is the same temperature difference (31°C) between ambient and air outlet temperature as is achieved for an ambient temperature of 35°C.

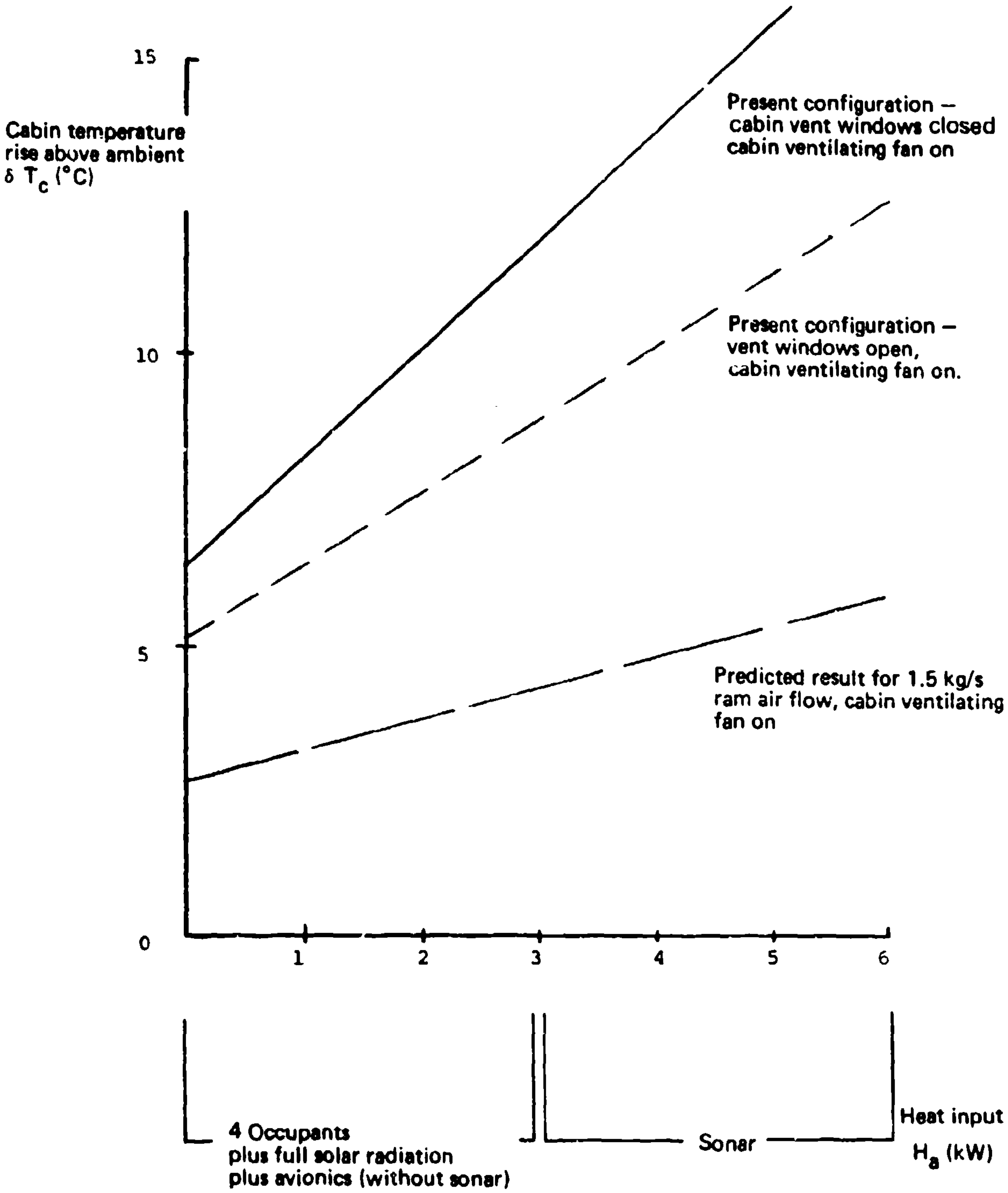

FIG. 5.1 CABIN TEMPERATURE RISE ABOVE AMBIENT AS A FUNCTION OF IMPOSED HEAT LOADING

The heat removed by the cooling air will be

$$h_c = m_c\, C_p\, (T_c - T_{\text{inc}}), \tag{6.1}$$

Where h_c = heat removed (kW);

T_{inc} = inlet temperature of cooling air (°C);

m_c = cooling air mass flow (kg/s);

C_p = specific heat of air (kJ/kg °C),

= 1·0 kJ/kg °C.

Then, from equation (A1.20), and assuming that there are no leakage airflows and the cabin ventilation fan is off, and including the heat removal term h_c above,

$$H_a = 0{\cdot}17\,(T_c - T_a) + m_c\,(T_c - T_{\text{inc}}) - 0{\cdot}98. \tag{6.2}$$

Thus, with air leaving the cooling unit at 4°C (case (*a*)) equation (6.2) becomes, after re-arrangement,

$$T_c = 3{\cdot}54\, H_a + 0{\cdot}6\, T_a + 5{\cdot}1. \tag{6.3}$$

If there exists a temperature difference of 31°C between ambient and cooling air outlet temperature (case (*b*)), then for an ambient temperature of 43°C, cooling air outlet temperature is 12°C. Then, equation (6.2) becomes

$$T_c = 3{\cdot}54\, H_a + 34. \tag{6.4}$$

Equations (6.3) and (6.4) are presented graphically in Figure 6.1; several values of T_a have been substituted into equation (6.3).

The resulting WBGTs are calculated according to equation (2.1) and are plotted in Figure 6.2. These values are based upon the air leaving the cooling unit in a saturated condition, resulting in a moisture content of

0·005 kg moisture/kg dry air (outlet 4°C),
0·009 kg moisture/kg dry air (outlet 12°C).

6.2 Other Alternatives

Performance details of possible alternatives to the NGL system are not available at the present time. There are proposals for Harrison (vapour cycle) cooling units, and in view of the marginal performance of the NGL Commando cooling unit in the Sea King these other proposals should be examined.

6.3 Discussion

In Section 6.1 it has been shown that the NGL cooling system could provide a marginally satisfactory cabin environment (as defined by a WBGT index of 28°C), if a 4°C cooling unit outlet temperature could be reached in high ambient temperature conditions (43°C), and all of the Sonar heat is ducted overboard. There are, however, a number of limitations associated with operation of this cooling unit. These are—

(*a*) No reserve cooling capacity for reducing cabin temperature after the aircraft has been "soaking" in full solar radiation.

(*b*) The cooling unit derives its effectiveness largely from *drying* the cabin air—resulting in a cabin relative humidity of approximately 10%. While physiological ill-effects will not normally result from short exposures to this low relative humidity (these humidity levels are common in commercial passenger carrying aircraft (Human Engineering Group, ARL, 1965—Ref. 9)) rapid dehydration of the crew members will occur. It would be essential then for crew members to be well hydrated prior to flight and for intake of liquids to be provided during flight.

(*c*) There would be an appreciable time lag—as derived in Appendix 2 (see also Figure 6.3), before cabin conditions attain the acceptable level. This effect will be accentuated if the aircraft has been subject to a "hot soak".

(*d*) In view of this time lag, the possibility of using sun-shields for the transparencies, and/or ground based cooling units should be considered in conjunction with the NGL Commando cooling unit because of its marginal capability.

Again, airflow visualisation studies on a model or in an actual cabin would be very useful, to maximise the effectiveness of any cooling unit, since the position and direction of inlets *and* outlets can play an important part in making cooling effective.

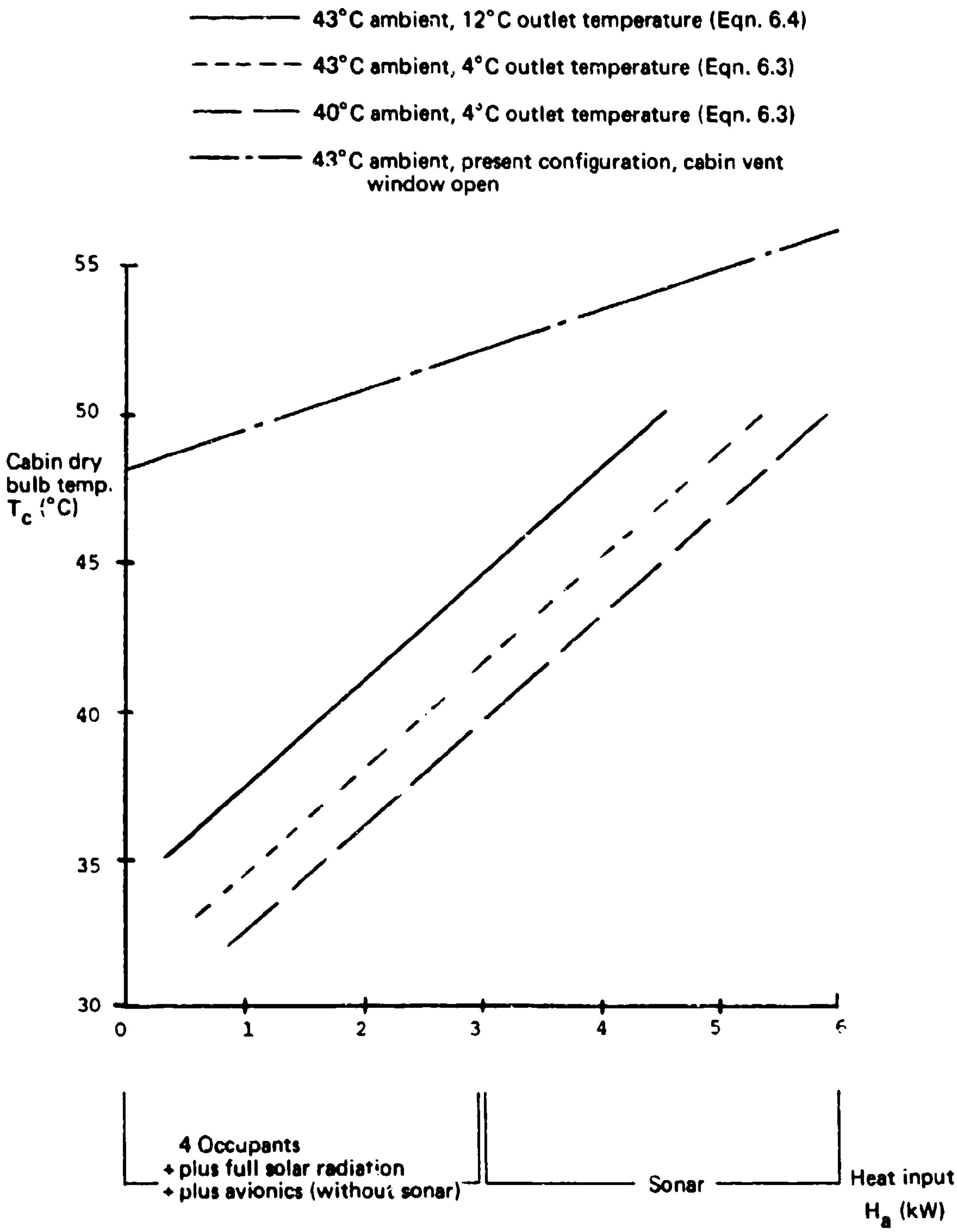

FIG. 6.1 PREDICTED CABIN TEMPERATURE AS A FUNCTION OF IMPOSED HEAT LOADING FOR SEA KING WITH NGL COOLING UNIT.

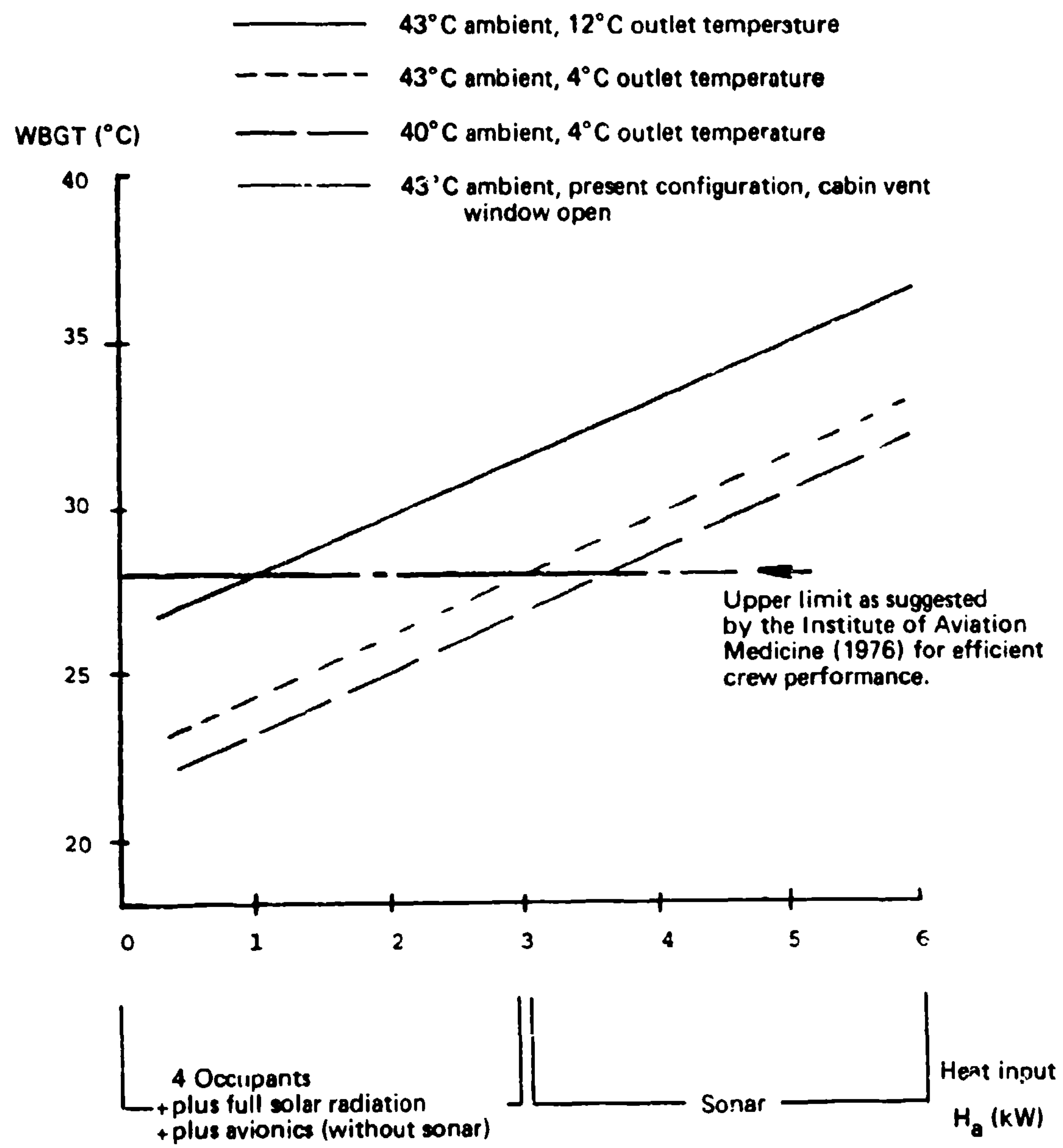

FIG. 6.2 PREDICTED CABIN WET BULB GLOBE TEMPERATURE AS A FUNCTION OF IMPOSED HEAT LOADING FOR SEA KING WITH NGL COOLING UNIT.

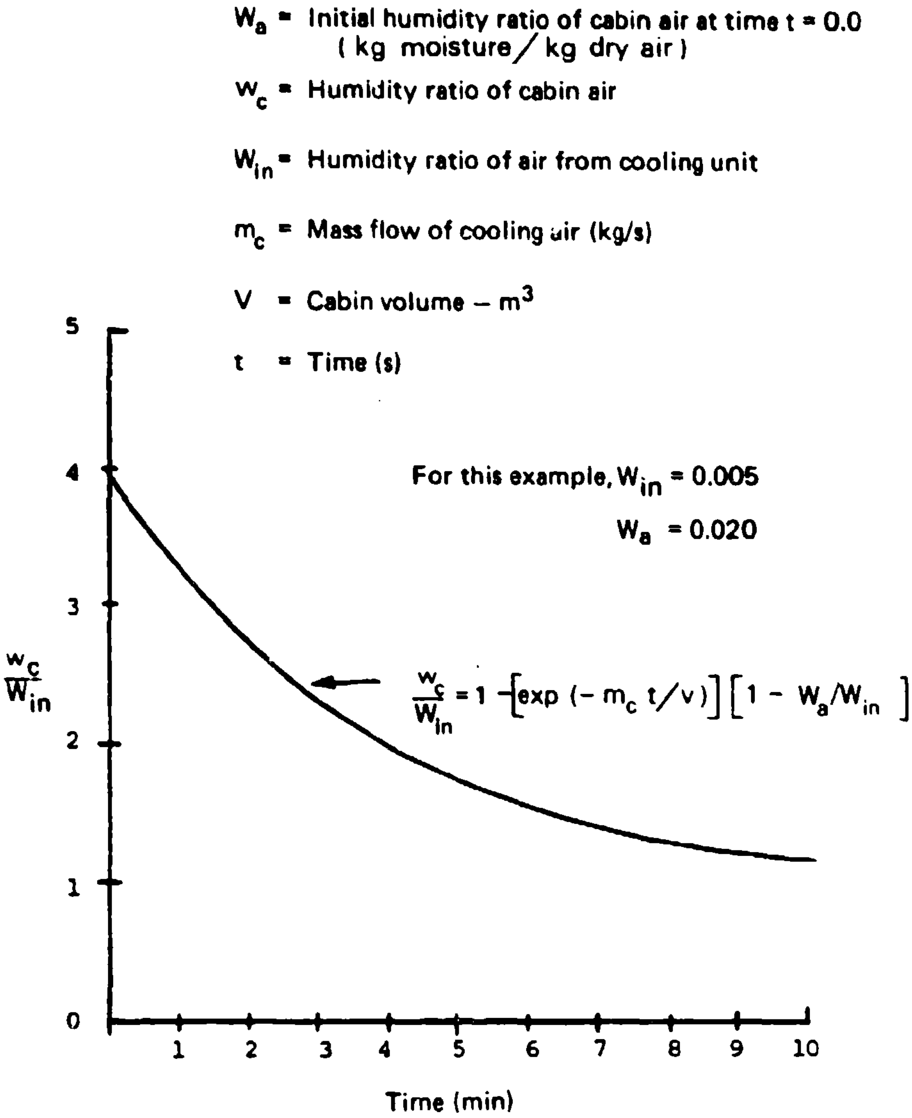

FIG. 6.3 MOISTURE CONTENT OF CABIN AIR AS A FUNCTION OF TIME.

7. CONCLUSIONS

7.1 Partial Alleviation of Heating Problem

There are several relatively simple short-term modifications to the aircraft which will provide much-needed partial alleviation of the heating problems. These modifications could conceivably be carried out before the Australian summer of 1977–78, and take the form of

(*a*) Ram air scoops—as manufactured by Sikorsky but with adequate air exit points to allow a through-flow of air,

(*b*) elimination of hot-air entry.

If feasible, a reduction in the temperature rise of cabin ventilation air (if ram-air cooling is not adopted), and elimination of at least some of the dissipation of electrical heat to the cabin should be undertaken, e.g. by using the present ventilating fan as an exhauster. The air surrounding the observer and sonar operator would then, however, be still, whereas at present a small flow is available from two "eyeball" vents. Thus it may be necessary to provide small individual fans; this has been carried out by Sikorsky (Directorate of Naval Engineering—Ref. 7).

7.2 Cabin Air Conditioning

An assessment of the NGL Commando cooling system proposal has shown that, provided that the outlet from the cooling unit can be maintained at 4°C or less at high ambient temperature, and sonar electrical heat is ducted overboard, then the system will maintain the cabin at a 28°C WBGT index level.

The limitations of this proposal are outlined in Section 6.3. The performance of this cooling unit is marginal, and its effectiveness derives mainly from its effect of producing very dry air, rather than from its effectiveness in reducing cabin temperature (the resulting cabin temperature will be only 1·5°C below an ambient of 43°C, without sonar equipment heating of the cabin).

The cabin environment will, however, be much improved compared with the present situation, or even the results obtained using ram-air ventilation of the cabin.

The provision of some protection from solar radiation prior to flight would be desirable in either case.

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APPENDIX 1
HEAT BALANCE OF SEA KING

A1.1 Theoretical Heat Flow Equation (Steady State)

The heat balance equation is based in part upon experimental observations at one particular cabin temperature, and in part on theoretical predictions. The theoretical heat balance for the aircraft in its current configuration, with vent windows and cargo door closed, is expressed by: Heat input from all sources = Heat removal by convection/conduction from cabin walls + heat removed by fan ventilating air + heat removed by other "leakage" airflows entering and leaving the cabin—i.e.

$$H_e + H_s + H_m + H_{eng} = K_s(T_c - T_a) + m_f C_p (T_c - T_{in}) + m_l C_p (T_c - T_a) \qquad \text{(A1.1)}$$

where H_e = cockpit and cabin electrical load (kW),

H_s = solar heat load (kW),

H_m = metabolic heat load from occupants (kW),

H_{eng} = heat input to cabin due to engine (including direct conduction, radiation and exhaust heating of skin) (kW),

T_c = cabin temperature (°C),

T_a = ambient air temperature (°C),

K_s = overall thermal conductivity of fuselage (kW/°C),

T_{in} = temperature of cabin ventilation air at point of entry to cabin (°C),

m_f = cabin ventilation fan air mass flow (kg/s),

m_l = leakage inflow of ambient air (kg/s),

C_p = specific heat of air (kJ/kg°C), taken here as 1·0.

Note that equation (A1.1) takes no account of additional inflows of hot air from the engine bay, as discussed in Section 3.1, because these occur only when front vent windows are open. If such flows are present, then the additional heat inputs are similarly calculable, provided the flow rate and temperature are known, from the equation.

$$H_d = m_d C_p (T_c - T_d), \qquad \text{(A1.2)}$$

where H_d = heat input due to hot air,

m_d = mass flow of hot air,

T_d = hot air temperature.

Rewriting equation A1.1:

$$H_a = \delta T_c (K_s + m_l + m_f) - \delta T_{vent} m_f - H_{eng}, \qquad \text{(A1.3)}$$

where $\quad H_a = H_e + H_s + H_m$ (known heat loadings), (A1.4)

and $\quad \delta T_c = T_c - T_a$. (A1.5)

$$\delta T_{vent} = T_{in} - T_a \qquad \text{(A1.6)}$$

= temperature rise in air passing through the ventilating fan.

A1.2 Thermal Conductivity of Fuselage

The value of K_s, the fuselage thermal conductivity, is assessed in part by theoretical considerations, and in part by reference to internal surface temperature measurements, which for one particular flight were as follows in Table A1.1.

TABLE A1.1

Temperatures in Sea King cabin during flight

Location	Temperature (°C)
Outside air temperature	16
Cabin air temperature	25
Surface of roof lining	25
Surface of insulation on cyclic pitch control duct	27·5
Surface of floor	19–21·5
Surface of rear cargo door	23
Surface of forward access door	19

The surface temperatures were measured by a standard thermocouple; a contact pyrometer not being available. Thus they may be slightly in error, also *local* air temperatures were not recorded. The main purpose of these measurements was to find if significant heating occurred due to the engine, and also to obtain an estimate of heat transfer through the floor.

The lack of variation in the roof lining temperature indicated that there is not a large heating effect due to the engine, although the slightly higher temperature of the bag insulation over the cyclic control duct indicates that there is a small heat input in this area.

The cabin walls may be considered in five sections;

(*a*) transparencies (area 7·7 m²);

(*b*) sides and roof (insulated) (28·7 m²);

(*c*) uninsulated forward door (1·6 m²);

(*d*) partially insulated cargo door (1·7 m²);

(*e*) rear bulkhead (3·4 m²) (designated a "thermal barrier")

Identical internal convective heat transfer coefficients of 8·3 W/m²°C (CSIRO, 1972—Ref. 10) from cabin air to the inside surface of all these sections, and a very high heat transfer coefficient for the exterior skin (where the air velocity is high), will be assumed. The heat flows through the sections are computed as follows:

(*a*) Transparencies—Natural convection inside, conduction through 5 mm thick plastic, conductivity 0·026 kW/m²°C, heat transfer = 0·049 kW/°C.

(*b*) Cabin sides and roof—These are lined with 50 mm of fibreglass insulation, conductivity 0·001 kW/m²°C, thus overall heat transfer = 0·028 kW/°C.

(*c*) Uninsulated forward door—Heat transfer = 0·013 kW/°C.

(*d*) Partially insulated rear door—Conductivity 0·003 kW/m²°C, thus heat transfer = 0·005 kW/°C.

(*e*) Rear bulkhead "thermal barrier"—Insulation of this is 25 mm fibreglass, 0·002 kW/m²°C.

Thus heat transfer = 0·006 kW/°C.

In-flight measurements of air temperatures in the tail section (just aft of the "thermal barrier") have shown an air temperature of 8°C above ambient. Thus, the heat transfer by conduction through the cabin walls (h_{wall}) is given by

$$h_{wall} = 0{\cdot}094\,(T_a - T_c) + 0{\cdot}006\,(T_a - T_c + 8), \qquad \text{(A1.7)}$$

hence

$$h_{wall} = 0{\cdot}100\,(T_a - T_c) + 0{\cdot}046 \qquad \text{(A1.8)}$$

(f) Heat flow through cabin floor—The heat flow through the cabin floor is calculated from the surface temperature measurements of Table A1.1. If a film convective heat transfer coefficient of 8·3 W/m²°C is assumed for the floor, then the heat transfer through to the floor is given by

$$h_f = 0{\cdot}0083\, A_f (T_c - T_f), \qquad \text{(A1.9)}$$

where h_f = heat transfer through the floor (kW),

A_f = floor area (15 m²),

T_f = floor temperature (°C).

The fuel (total capacity 3000 litres), is carried in tanks which extend under most of the cabin floor area. The floor temperature measurements were constant from the centre of the floor until within 100 mm of the cabin sides, thus indicating that the heat being conducted through the floor is probably transferred directly to the fuel, unless a high thermal conductivity path exists from the whole of the floor area to the outside of the fuselage.

Inspection of a Sea King airframe showed that this is unlikely. It appears reasonable, therefore, to consider that heat passes out of the cabin through the floor structure to the fuel; it is also probable that this fuel is initially at sea level ambient air temperature. As it has been calculated that a heat input to 3000 litres of fuel (full fuel load) of 1·3 kW hr is required to raise its temperature by one degree celsius, a constant fuel temperature equal to ambient air temperature may be assumed without significant error. The effects of partial tank emptying on heat conduction are unknown but probably small. On this basis, therefore,

$$h_f = A_f k_f (T_c - T_a) \qquad \text{(A1.10)}$$

$$= K_f (T_c - T_a) \qquad \text{(A1.11)}$$

Hence

$$K_f = 0{\cdot}0083 \times \frac{15\,(T_c - T_f)}{(T_c - T_a)} \qquad \text{(A1.12)}$$

then substituting known values of T_c, T_f, and T_a from Table A1.1,

$$K_f = 0{\cdot}069 \text{ kW/°C}. \qquad \text{(A1.13)}$$

(g) Overall cabin heat flows—Combining equations (A1.8) and (A1.13), we obtain

$$H_{cabin} = 0{\cdot}17\,(T_a - T_c)\,.\,(\text{kW}) \qquad \text{(A1.14)}$$

In this expression, the small constant heat flow in equation (A1.8) has been neglected as it is equivalent to only about 0·2°C cabin temperature differential.

A1.3 Leakage Air Flow

The leakage airflow of ambient air into the cabin is not reliably known; measurements have indicated that up to 0·11 kg/s (14·6 lbm/min) may leak in around the doors.

A1.4 Heating Effect Due to Engine

Equation (A1.1) may be rewritten in the form

$$H_{eng} = (K_s + m_f)(T_c - T_a) - m_l (T_c - T_a) - (T_{in} - T_a) m_f - (H_e + H_s + H_m). \qquad \text{(A1.15)}$$

Of these terms, K_s, m_f, $(T_{in} - T_a)$, are substantially constant (with the fan in operation). H_l, H_s, and H_m are calculable for particular flight conditions, and $(T_c - T_a)$ may be measured in flight. Since it is difficult to measure m_l accurately, there is a similar uncertainty in H_{eng}.

In one test flight

T_a = 16°C $\qquad m_f$ = 0·26 kg/s

T_c = 25°C $\qquad m_l$ = 0·11 kg/s

T_{in} = 25·5°C

H_m = 0·5 kW

H_e = 0·5 kW (sonar off)

H_s = 0·4 kW (estimate, based on cloud cover, time of day and season).

Substituting in equation (A1.15) above,

$$H_{eng} = 9\,M_l - 0{\cdot}01. \qquad \text{(A1.16)}$$

Hence, for the measured value of m_l

$$H_{eng} = 0{\cdot}98 \text{ kW} \tag{A1.17}$$

In the absence of evidence to the contrary, it must be assumed that this value is constant, and that alteration of m_l, e.g. by better door sealing, will result in a change in T_c.

A1.5 Heat Balance Equation

The final heat balance equation for the current Sea King is then

$$H_a = 0{\cdot}54\, \delta T_c - 3{\cdot}45. \tag{A1.18}$$

Equation (A1.18) is for the Sea King in its current configuration, with front vent windows closed and cabin air ventilation fan in operation. By reference to the more general form of the heat balance equation (A1.3), where $H_a = \delta T_c (K_t + m_l + m_f) - \delta T_{vent}\, m_f - H_{eng}$, the effect of changes in the parameters (as outlined in Section 4) can be assessed. Thus by substitution of K_t from Equation (A1.14) and H_{eng} from Equation (A1.17), there results

$$H_a = \delta T_c (0{\cdot}17 + m_l + m_f) - \delta T_{vent}\, m_f - 0{\cdot}98. \tag{A1.19}$$

The incorporation of ram air cooling will bring equation (A1.19) to the form

$$H_a = \delta T_c (0{\cdot}17 + m_l + m_f + m_r) - \delta T_{vent}\, m_f - 0{\cdot}98 - \delta T_r\, m_r \tag{A1.20}$$

where m_r = ram air mass flow (kg/s),

δT_r = stagnation temperature rise of ram air (°C).

APPENDIX 2
MOISTURE CONTENT OF CABIN AIR AS A FUNCTION OF TIME

The air conditioning unit will in general supply air to the cabin, at a moisture content different from that existing in the cabin. For example, at start-up the cabin has been open to ambient outside air.

Because of the small mass flow rate of the proposed cooling unit, an appreciable time lapse can be expected for the cabin conditions to attain the moisture content of the "cooling" air.

Now, taking V = cabin volume (m^3),

m_c = cooling air mass flow (kg/s),

W_{in} = humidity ratio of incoming air (kg moisture/kg dry air),

w_c = humidity ratio of cabin air (kg moisture/kg dry air),

t = time (seconds),

W_a = humidity ratio of ambient air,

and assuming that the moisture content of exit air is equal to the moisture content of the cabin air, and that the moisture content of the cabin is distributed evenly throughout, then, the rate of change of cabin humidity ratio,

$$dw_c/dt = (m_c W_{in} - m_c w_c)/V. \tag{A2.1}$$

$$dt = (V/m_c)\int 1/(W_{in} - w_c)\, dw_c, \tag{A2.2}$$

$$t = -(V/m_c)\, ln\, (W_{in} - w_c) + C, \tag{A2.3}$$

where C is a constant, determined from initial conditions. Equation (A2.3) can be reformed to

$$C' \exp(-m_c t/V)^* = W_{in} - w_c \tag{A2.4}$$

then

$$w_c = W_{in} - C' \exp(-m_c t/V)$$

* exp (u) is taken here to be e^u.

and as at time $t = 0$,

then $$w_c = W_a,$$

$$C' = W_{in} - W_a,$$

$$w_c = W_{in}[1 - \exp(-m_c t/V)] + W_a \exp(-m_c t/V)$$

and

$$w_c/W_{in} = 1 - \exp(-m_c t/V) + (W_a/W_{in}) \exp(-m_c t/V)$$
$$= [\exp(-m_c t/V)][W_a/W_{in} - 1] + 1.$$

The ratio w_c/W_{in} is plotted versus time in Figure 6.3.

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